ENGINEER'S notebook

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SYSTEM DESIGN STUDY USING DYNAMIC SIMULATION OF A PROPYLENE REFRIGERATION PROCESS COMPRESSION TRAIN

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INTRODUCTION

The procurement of the compressor and driver included a dynamic simulation study of the system. The compressor vendor was the prime contractor to supply the rotating equipment and perform the dynamic simulation study. The simulation established the requirements for the prevention of surge during normal operation and emergency shutdown. The study also evaluated the parameters to start the compressor train based on the client's specified operational constraints.

This paper describes the process and equipment along with the initial simulation study results. The study indicated that modifications to the process and equipment were beneficial.

THE PROCESS

The process uses a two-section refrigeration cycle using propylene as the heat transfer medium. The process heats a low-temperature ethylene stream. The process is shown on the Pressure-Enthalpy diagram (Figure 1). Starting from the compressor final discharge the hot high-pressure gas flows to the main condenser where it is cooled and condensed to a liquid. The liquid then flows to the second-stage ethylene heat exchanger where the propylene is further cooled. The propylene liquid pressure is dropped through a valve to the second-stage compressor inlet pressure. The two-phase mixture flows through a separator where the gas stream flows to the compressor sidestream (interstage) and the separated liquid flows to a liquid holding tank. From the holding tank the liquid flows to a first stage heat exchanger. The cooler exit stream flows through a valve that drops the stream pressure to the first-stage compressor inlet pressure. The two-phase fluid flows to a separator. The stream temperature is controlled by the addition of hot discharge gas from a recycle valve mixing with the cooled two-phase fluid. The gas flow out of the separator goes to the first section compressor inlet. Any liquid in the separator is pumped back to the final discharge holding tank (accumulator).

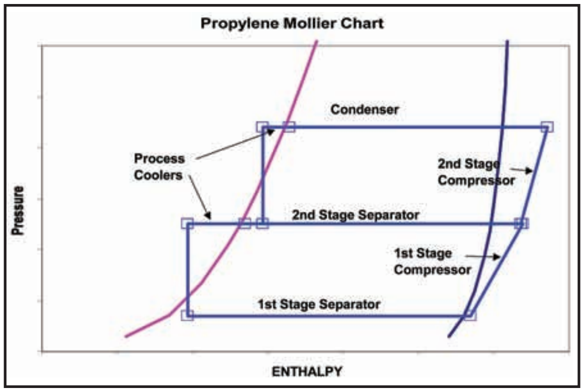

Figure 1. Propylene refrigeration cycle pressure-enthalpy chart.

SITE CONDITIONS

The plant is located in the southwest United States.

- Ambient temperature ranges from –20 to 100 ^{O}F
- Normal barometric pressure is 12.8 psia
- Site elevation of 3600 ft.

CLIENT PROCESS CONSTRAINTS

The client specified initial system constraints for the refrigeration compression system.

1. There was no onsite storage of propylene refrigerant. This encourages a constraint to minimize the loss of process fluid. On a unit shutdown, the system should not be blown

ABSTRACT

This paper discusses knowledge gained during the installation design of a propylene refrigeration process compression train using a dynamic simulation model. The discussion will follow the steps taken from initial conceptual design through modifications based on dynamic simulation of the selected equipment and process. The original process control system and requirements for start up and shut down are discussed. The dynamic simulation results of the original design and the subsequent redesign of the system are presented.

The propylene refrigeration system has a centrifugal compressor driven by an electric motor with a VFD control and a speed-increasing gearbox. The system was evaluated using a dynamic process simulation model to size valves and determine the suitability of system components. Through a collaboration of efforts of several people, revisions to the original system were implemented. The final result was a successful plant start up of this equipment without motor overload or compressor surge in July 2007.

down. Refrigerant replacement would have to be done by tanker truck. The nearest supply source was located 300 miles from the site.

2. The process does not have an inlet throttle valve before the first compressor stage.

3. The compressor had to start with a settle-out temperature of 100 ^{O}F in the fluid system. [Note: Since there was no isolation of the liquid and gas sections, at 100 ^{O}F some of the liquid will vaporize (boil off) and the equilibrium system pressure will be 227 psia. This pressure exceeds the normal compressor discharge pressure of 215 psia. See the Pressure-Enthalpy diagram (Figure 2) and the P-T diagram (Figure 3, Ref. 4).]

4. The system design needs to prevent or minimize compressor surge on an emergency trip.

5. The system design needs to prevent compressor surge during start up.

COMPRESSOR TRAIN EQUIPMENT

The compressor train consists of an induction motor, speed increasing gear, and a two-section centrifugal compressor.

The driver is an 1800 rpm, ~10000 hp induction motor with a variable frequency drive (VFD) system. The motor was cooled by internal shaft-mounted fans.

The gear is a speed increasing gear with a speed ratio of 3.127.

The compressor was a two-section centrifugal with a single sidestream inlet with operating speed at 5778 rpm. The first section of compression has a suction pressure of ~40 psia; the second section has a suction pressure of ~70 psia and a discharge pressure of ~215 psia.

SIMULATION PROCESS MODEL

The client supplied the following information from which the simulation model was created.

- Process and instrumentation diagrams
- Process flow diagrams and supporting data
- System volumes of piping, vessels, and coolers
- System process control philosophy and methodology

The process was modeled using the QMC Dyflo dynamic simulation program (Ref. 1.) in creating the dynamic simulation model. The initial model is shown on Figure 4. The main inlet, sidestream inlet, and discharge volumes for piping and vessels were lumped as a single volume at each section. The pressure drops were accounted for by using valves. Not all of the components were modeled on the liquid side of the process.

The sidestream connection was modeled such

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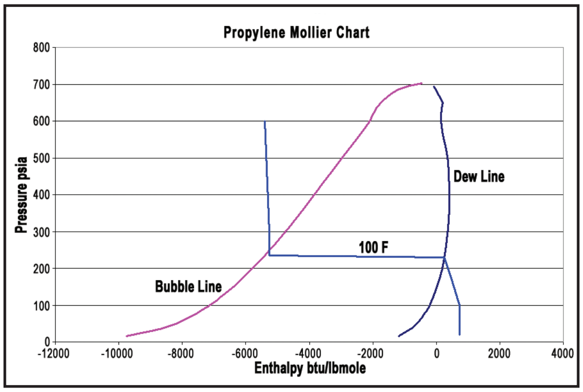

Figure 2. Propylene pressure enthalpy chart.

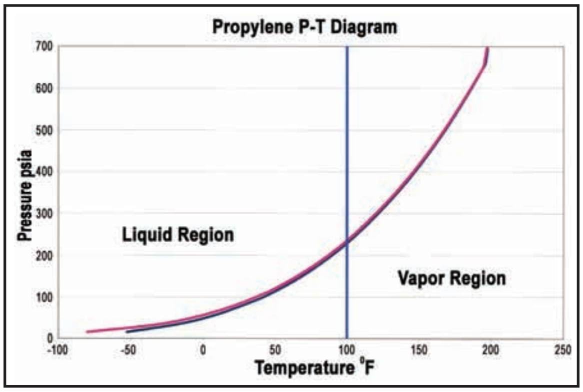

Figure 3. Propylene pressure temperature chart depicting liquid - gas phase regions.

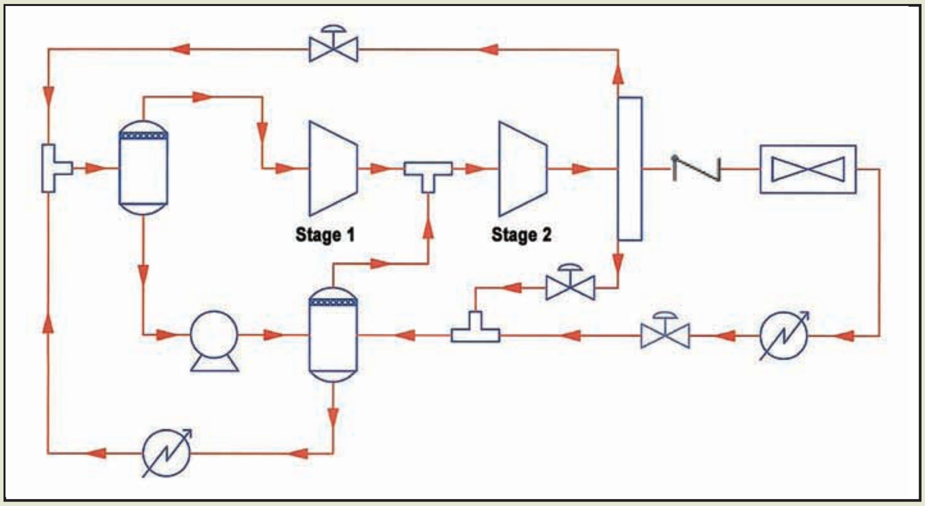

Figure 4. Propylene process schematic initial design.

that the pressure changes inside the compressor did not instantaneously affect the external stream pressure. The compressor internal flow is at high velocity and the mixing of flow streams is based on static pressure, not total pressure. More information on the sidestream mixing is described in the ASME PTC-10 Compressor Test Code Ref. 3.

The DyFlo modified Lee-Kessler equation of state was used for the fluid properties (Ref. 2 patent #7,228,241).

Each compressor section flow, head, speed, and efficiency map was digitally characterized from predicted performance data. A proprietary calculation code was used to define compressor performance. This code uses the real gas properties for pressure, temperature, volume, and enthalpy to establish the head, flow, and efficiency parameters during the transient calculations.

SURGE EVALUATION AND RELATED INITIATIVES

Once the model was completed, the simulation was run for steady state to validate the model code. The anti-surge valves were sized using standard design rules (Ref. 5). The model was then exercised to simulate an emergency shutdown event from the specified design operating point. This was

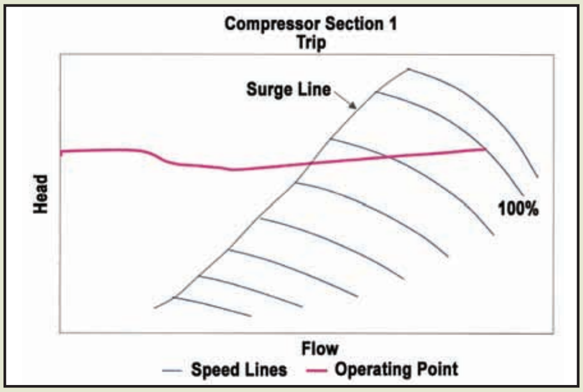

Figure 5. Compressor section 1 head vs. flow response to ESD event with initial system design.

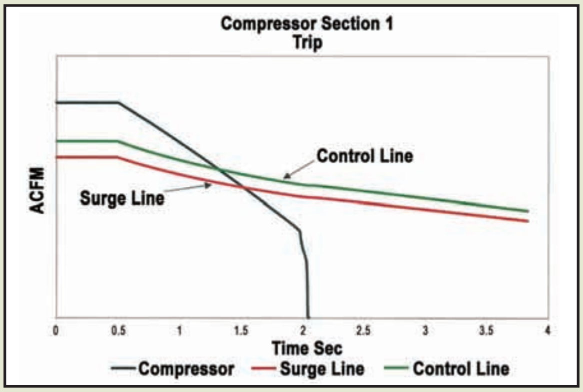

Figure 7. Compressor section 1 flow vs. time response to ESD event with initial system design.

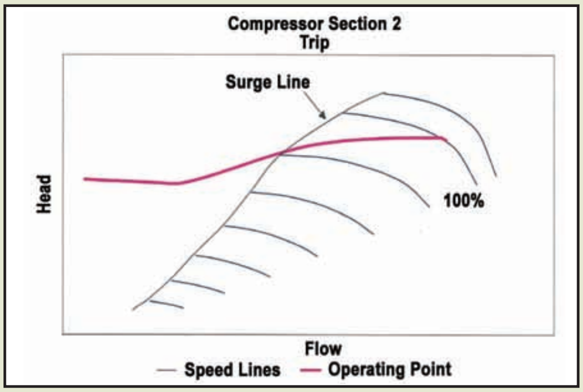

Figure 6. Compressor section 2 head vs. flow response to ESD event with initial system design.

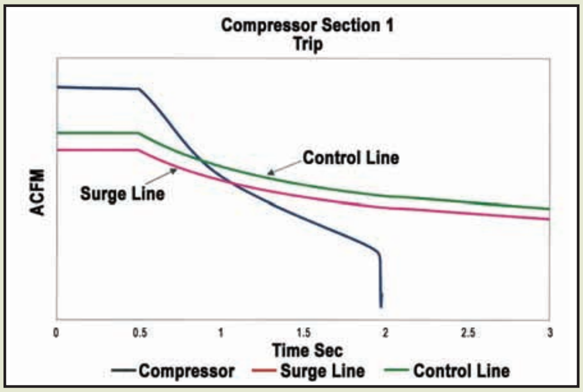

Figure 8. Compressor section 2 flow vs. time response to ESD event with initial system design.

used to confirm the anti-surge valve sizing and system response to the transient event.

Initial runs indicated the unit could not avoid surge with the client-supplied process arrangement. Figures 5 and 6 show a result of the initial run on the head flow map for each section. Figures 7 and 8 show the event as flow versus time. The sidestream pressure was essentially constant due to the large connected volume. The main inlet volume was also very large, therefore, the inlet pressure also remained constant. This resulted in section 1 external pressure ratio and head to remain essentially constant as the compressor decelerated. Thus, the first section compressor could not generate the required head during deceleration to match the system head, so back-flow occurred, resulting in compressor section surge. The second compressor section also went into surge. This is typical of many centrifugal compressors with sidestream feeds.

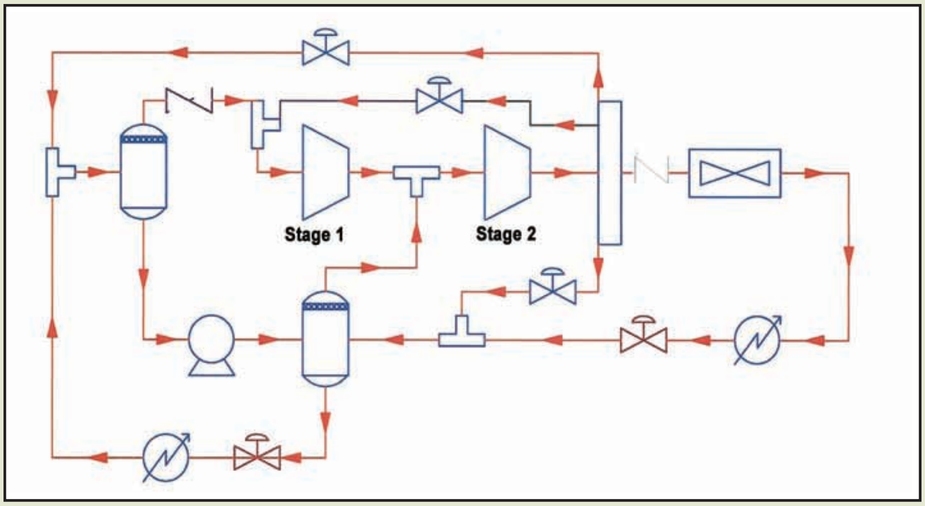

Figure 9. Propylene process schematic revised design for trip case.

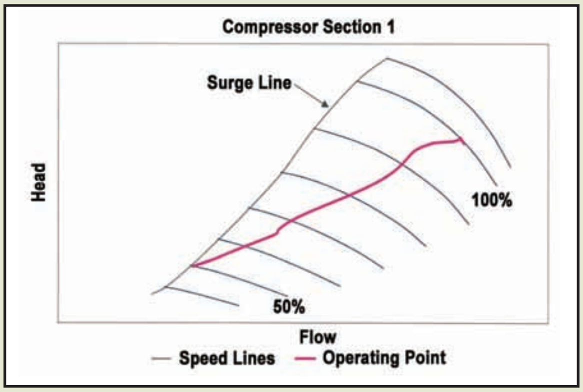

Figure 10 Compressor section 1 head vs. flow response to ESD event with revised system design.

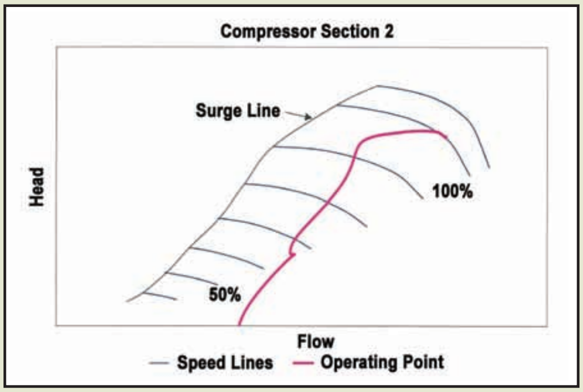

Figure 11. Compressor section 2 head vs. flow response to ESD event with revised system design.

To resolve the compressor surge-on-trip dilemma, modifications to the process piping were proposed. An additional recycle line from final discharge to compressor inlet was modeled. A check valve was added between the separator and the compressor inlet flange. The new recycle line was tied into the inlet piping between the compressor inlet flange and the check valve. This effectively reduced the system suction volume for the trip condition.

Numerous simulation runs were made to identify an acceptable inlet volume and emergency shutdown valve (ESD) capacity, characteristic, and stroking speed. The concept of reducing the inlet volume was to allow the inlet pressure to rise during a trip event. Also, the inlet temperature would rise since it would not be cooled by the large inventory of fluid on the inlet. Consequently, the suction pressure rise reduced the first section external head requirement and the temperature increase allowed an additional increase in volume flow.

The resulting model is shown in Figure 9. Upon trip, the discharge check valve prevented back-flow from the condenser section. As flow develops across the ESD valve, the isolated first-section inlet pressure will rise. The discharge recycle valve also will open upon an emergency trip to dump the discharge section mass inventory and lower the discharge pressure.

The final simulation demonstrates the unit decelerated without either section going into surge in the normal operating speed range. A point near surge was used as the system design worst case. Figures 10 and 11 show the head flow map for each compressor section. Section 1 heads towards surge at low speed and hence low energy. A further discussion on this issue is available in Ref. 5. The flow vs. time plot with the predicted surge limit line and surge

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control line are in Figures 12 and 13. The unit deceleration is shown in Figure 14.

START UP

The start up at a settle-out temperature of 100 ^{O}F would occur on a long-term shutdown because the refrigeration loop components are all insulated. As referenced earlier, Figure 3 is a plot of the propylene P-T diagram showing the bubble and dew point temperatures. Based on these ambient conditions, the system will have an equilibrium pressure of 227 psia. This is above the compressor discharge pressure at system design of ~215 psia.

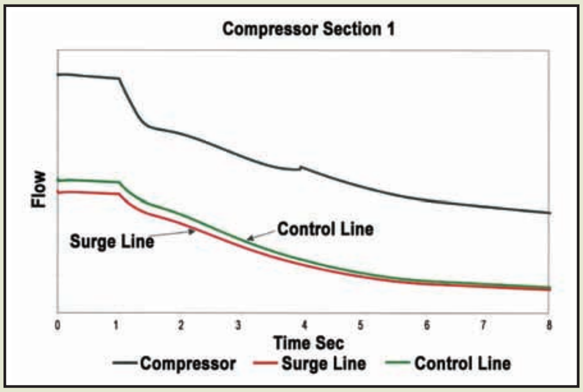

Figure 12 Compressor section 1 flow vs. time response to ESD event with revised system design.

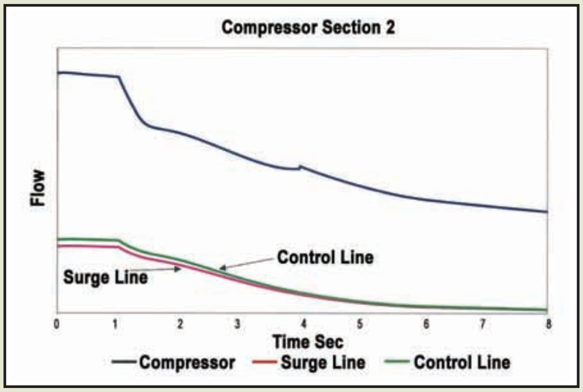

Figure 13. Compressor section 2 flow vs. time response to ESD event with revised system design.

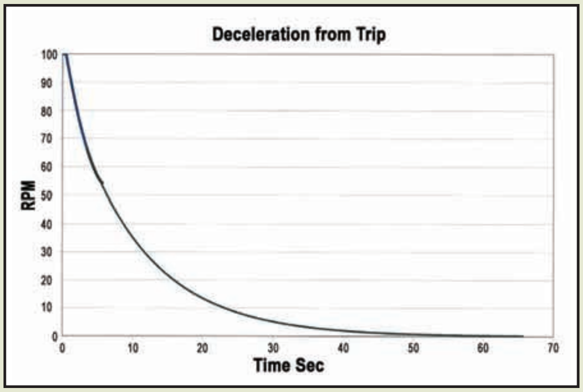

Figure 14. Train deceleration RPM vs time for ESD event.

The simulation quickly revealed that a start up from this settle-out pressure would require a much larger motor and would also exceed the discharge system components maximum working pressures. The initial start-up runs indicated problems with the motor/VFD system and the overall process piping arrangement.

After initial evaluation the following three improvements were proposed and investigated

- Revise the motor design to utilize full capacity of the VFD system
- Investigate blow-down (even though blow-down is undesirable)
- Investigate piping changes to start up without blow-down of the refrigerant

MOTOR

INVESTIGATION

The induction motor as proposed had integral shaft cooling fans. This cooling scheme inhibited the motor's use of the full capacity of the VFD control system. The VFD can supply power to the motor that would allow it to produce full torque from 25 percent speed to 100 percent speed with the application of full-load current to the stator windings (Figure 15).

At this stage of the design and procurement, the motor could be modified. The motor design was changed to use external air cooling to allow for full motor torque during start up. The external air cooling system requires an inlet filter, motor-driven fans, and associated ducting and controls. Because the proposed changes were defined early enough in the design phase,

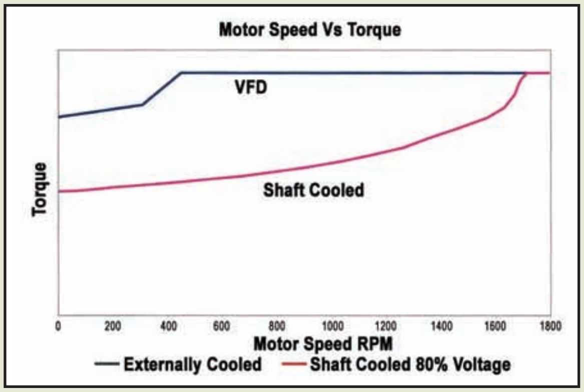

Figure 15. Motor speed vs.torque capability with integral fan and external ventilation design.

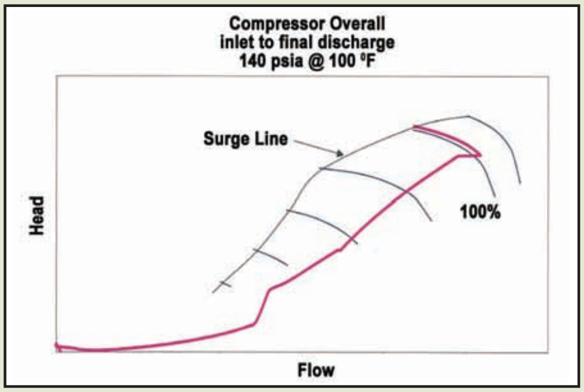

Figure 16. Compressor overall head vs. flow response to startup event at 140 PSIA and 100 degrees day.

modifications to the components, as well as the facility layout, could be accommodated.

BLOW-DOWN INVESTIGATION

Blow-down of the system to a lower settle-out pressure was considered but was not preferred if another solution could be found. As noted earlier, there was no provision for on-site storage of refrigerant. This would require several tanker trucks to be available to recharge the system once the compressor system was restarted. There also was no inlet throttle valve, which could be used to lower inlet pressure to the first compression stage as the unit is starting up.

By successive simulation runs, it was found that a system pressure of 140 psia at 100 ^{O}F allowed the compressor train to reach minimum operating speed. The simulation model predicted the system restart (Figure 16), the head flow map for the overall compressor. The acceleration of the train is shown in Figure 17. The motor speed torque curve and compressor load are shown in Figure 18.

An alternate system was needed to meet the client's and site installation requirements.

PIPING REVISIONS

After extensive review of the system process and instrumentation diagram it became apparent that, upon shutdown, the system had to separate the liquid and gas streams to minimize the settle-out pressure. The pressure in the original shut-in system rises because the liquid vaporizes into the gas phase. Modifications to the piping system required the addition of a new valve and the modification to the control of other valves. Figure 19 shows the new valve added to the process, and the existing valves that will automatically close when the unit is shut down. This effectively separates the gas from the liquid sections of the system. The resulting trapped gas volume will then settle out at 88 psia at 100 ^{O}F.

START UP 88 PSIA 100 ^{O}F

The simulation of this case with the revised motor and system piping was effective for the desired outcome. Figure 20 shows the compressor map for the overall compressor. Figure 21 shows the acceleration of the compressor train. Figure 22 shows the torque available from the motor and the required compressor load torque.

The system modifications to the motor and piping were implemented.

FIELD START UP

The compressor train was started for the first time in July 2007. The unit did not surge during acceleration up to speed as designed. Since starting, they have had some system shutdowns – all without reported compressor surge.

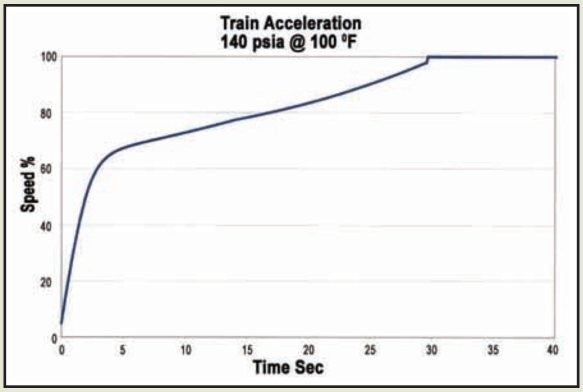

Figure 17. Train acceleration RPM vs time for startup event at 140 PSIA and 100 degrees day.

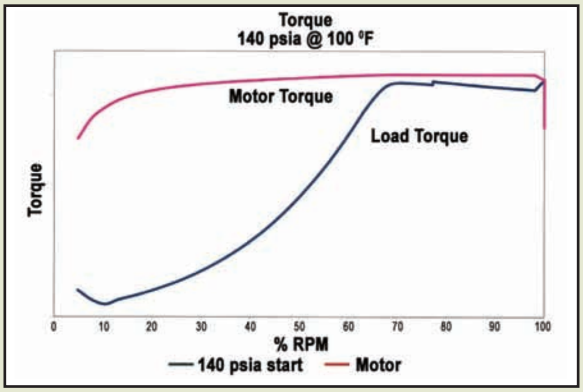

Figure 18. Torque vs. percent speed for startup at 140 PSIA and 100 degrees day motor and load.

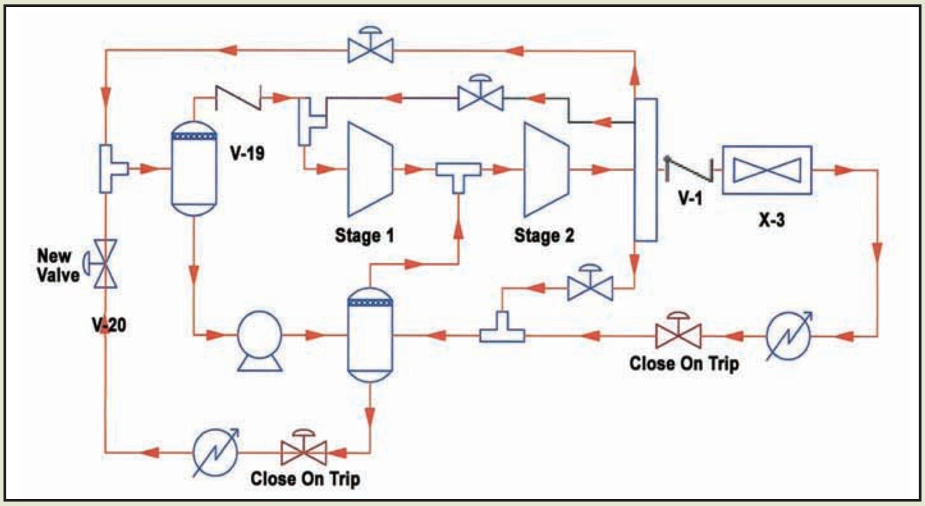

Figure 19. Propylene process schematic revised design for startup with reduced loop volume.

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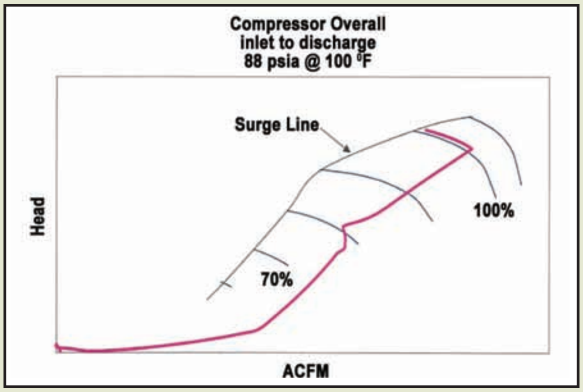

Figure 20. Compressor overall head vs. flow response to startup event at 88 PSIA and 100 degrees day.

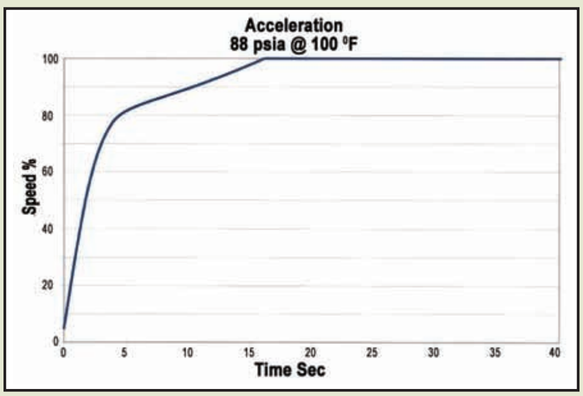

Figure 21. Train acceleration RPM vs time for startup event at 88 PSIA and 100 degrees day.

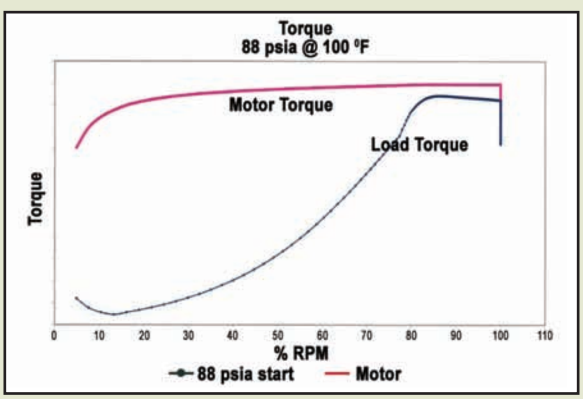

Figure 22. Torque vs. percent speed motor and load for startup at 88 PSIA and 100 degrees day.

CONCLUSIONS

A compressor system with side-streams can be designed to shutdown without surging and start up without surging. However, changes to conventional piping systems are prescribed.

- Identify problems with system components.
- Match the driver and compressor to the system.
- Demonstrate control system strategy, so modifications can be implemented if required.
- Size control valves to meet the total system requirements.
- When using a VFD for motor control, the motor should be designed with external cooling to utilize the full capability of the motor/VFD system when required.
- Many other system components, such as relief valves, blowdown valves etc., can have their designs confirmed by simulation allowing a successful system design.

Dynamic simulation is a design tool that can now be used to better assure system integration by testing various transient scenarios before committing to hardware.

NOMENCLATURE:

ESD – Emergency Shut Down
Head – Compressor Head

PF – Electrical Power Factor

Pressure – Pounds Per Square Inch Absolute (pressure) (psia)

RPM – Revolutions Per Minute (speed)

°F – Degrees Fahrenheit (temperature)

Lb.-Ft – Pound Foot (torque)

VFD – Variable Frequency Drive

REFERENCES:

1. DyFlo Instruction Manual, Quality Monitoring & Control Co., Kingwood, Texas

2. Patent # 7,228,241, "Systems, Methods And Apparatus For Determining Physical Properties Of Fluids"

3. ASME PTC-10-1997, Performance Test Code on Compressors, American Society of Mechanical Engineers, New York, New York

4. WinSim Design II Process Simulation Program V9.43, WinSim Inc., Houston, Texas

5. Application Guideline for Centrifugal Compressor Surge control Systems, Version 4.1, 2007, GMRC and Southwest Research Institute

ACKNOWLEDGEMENTS

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